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FFNo 672 Aug 65

Prepared for AIAA Meeting,
June 17-20, 1963
Los Angeles, California

X 63 12744

33 p

N65-88666

code-2A

A LOOK AT THE THERMODYNAMIC CHARACTERISTICS OF BRAYTON CYCLES FOR SPACE POWER

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INTRODUCTION

For a variety of space missions, both manned and unmanned, there exists a need for systems capable of generating power for many thousands of hours of continuous operation. Power requirements range from a few kilowatts for auxiliary use in near space missions up to many megawatts for manned missions utilizing electric propulsion to reach the other planets of our solar system. Powerplant specific weight (powerplant weight per kilowatt power output) for the high power level systems must be kept low because of (1) the large inherent size of these systems and/or (2) the strong dependence of electric rocket performance upon the powerplant weight.

The most promising power generation system for near future application to missions requiring power levels of several kilowatts or more is the indirect conversion closed loop system where heat is generated in a nuclear or solar source and rejected by a radiator, with power being obtained from a turbine located in the working fluid loop. The radiator has been shown to be the largest component and a major weight contributor to the powerplant and may easily constitute half or more of the total weight, especially at the high power levels. Much attention has been given to the vapor-liquid (Rankine) system using a metal working fluid since this system has a much better thermodynamic potential than a gas (Brayton) system for obtaining the low radiator specific areas and weights required for large power output applications. For comparable turbine inlet temperatures, a Brayton cycle radiator may require more than ten times the specific area (square feet per kilowatt) required for a Rankine cycle radiator (refs. 1 and 2). However, for non-propulsive power systems of several hundred kilowatts or less, where low specific weight is not so critical a requirement, the Brayton cycle merits consideration because its use eliminates (1) the problems associated with two phase flow in a zero gravity environment, (2) the presence of a severely corrosive working fluid, and (3) the possibility of erosion damage to the rotating components. Much of the required equipment and technology for the Brayton cycle is presently available, and this system has a good potential for multiple starts as well as for achieving the required long time reliability.

The thermodynamic characteristics of Brayton cycle space power systems have been discussed to a limited extent by a number of investigators (e.g., refs. 1 to 4). Each of these studies, however, was made for certain idealized or specific conditions and none of them considered all the pertinent system parameters. In view of these considerations, an analytical investigation was conducted in order to supplement the previous studies and obtain a better understanding of the thermodynamic characteristics of Brayton cycles for space application. This paper presents the results of the investigation.

SYMBOLS

A_i radiator internal heat transfer area

A_R radiating area

c_p specific heat, Btu/(lb)(^oR)

E recuperator effectiveness

[redacted]

h_i radiator gas film heat transfer coefficient based on internal heat transfer area, Btu/(hr)(sq ft)($^{\circ}$R)

h_R radiator gas film heat transfer coefficient based on radiating area, Btu/(hr)(sq ft)($^{\circ}$R)

Δh enthalpy change, Btu/lb

P shaft power, kw

p pressure, psia

r pressure ratio (greater than unity)

T temperature, $^{\circ}$R

w weight flow, lb/hr

γ specific heat ratio

ϵ emissivity

η efficiency

σ Stefan-Boltzmann constant, 0.173×10^{-8} Btu/(hr)(sq ft)($^{\circ}R^4$)

Subscripts:

C compressor

cy cycle

id ideal

s sink

T turbine

w wall

1 heat source exit or turbine inlet

2 turbine exit or recuperator inlet

3 recuperator exit or radiator inlet

4 radiator exit or compressor inlet

5 compressor exit or recuperator inlet

6 recuperator exit or heat source inlet

CYCLE ANALYSIS

A schematic diagram of a Brayton cycle configuration is shown in figure 1(a) and the corresponding temperature-entropy diagram in figure 1(b). The circled numbers correspond to the state point designations used in the analysis. The heat source exit gas at point 1 expands through the turbine to point 2, thereby producing the mechanical work necessary to drive the compressor and alternator. From the turbine, the gas enters the recuperator where it is cooled to point 3 as it transfers heat to the gas from the compressor. Final cooling of the gas to point 4 takes place in the radiator, where the excess heat is rejected to space. The gas leaving the radiator is then compressed to point 5, heated in the recuperator to point 6, and further heated back to point 1 in the heat source. Alternate configurations can include a liquid heating loop and/or a liquid cooling loop. However, the gas loop remains the same as described above except that heat exchangers replace the heat source and/or radiator.

The purpose of the cycle analysis was twofold: (1) to determine cycle performance, denoted by cycle efficiency, for any chosen set of cycle conditions, and (2) to select a set of cycle temperatures that is somehow advantageous to the system. Therefore, before the cycle analysis was started, some criterion had to be chosen to serve as a guide for the selection of desirable cycle temperatures. The most logical criterion seemed to be minimum system size and/or weight; however, such a minimization procedure would require an effort well beyond the scope of a preliminary study. Experience has shown that the radiator is (1) the largest component, by far, in a nuclear-powered system, (2) one of the two largest components in a solar-powered system, and (3) a major contributor to powerplant weight in both nuclear and solar systems. In addition, the size and weight of the radiator are greatly affected by cycle temperature selection. In a nuclear system, the weight of the reactor, the other major weight contributor, is affected to a much smaller extent than is the radiator by the selected cycle temperatures because the size of the reactor is determined primarily by nucleonics considerations. In a solar system, however, the size and weight of the collector, the other large component, are affected to large extent by cycle temperature selection, but not to the same extent as that of the radiator. Consequently, some aspect of radiator size appeared to be a logical criterion for cycle temperature selection. Experience has further shown that the size and weight of a fin and tube radiator are minimized at approximately the same cycle temperatures as are required to minimize the radiating surface of a prime area radiator. A prime area radiator, for the purposes of this study, can be defined as either a tubular radiator without fins or a tube and fin radiator with a 100 percent fin efficiency. Since prime radiating area is an indicator of both size and weight of the radiator, it was chosen as the criterion for cycle temperature selection. The cycle temperatures selected in this manner will be very near the optimum ones for nuclear systems and will deviate only slightly from the optimum ones for solar systems.

Cycle Efficiency

The cycle analysis was made using mechanical shaft power to the alternator as the basis. Computation of cycle efficiency was performed using the following assumptions.

(1) The working fluid is an ideal gas; consequently, specific heat is a constant independent of temperature.

(2) No heat losses from the system.

Cycle efficiency is defined as

$$\eta_{cy} = \frac{\text{Net shaft power}}{\text{Heat supplied}} = \frac{\text{Heat supplied - heat rejected}}{\text{Heat supplied}}$$

Symbolically, this definition is

$$\eta_{cy} = \frac{wc_p(T_1 - T_6) - wc_p(T_3 - T_4)}{wc_p(T_1 - T_6)} = \frac{T_1 - T_6 - T_3 + T_4}{T_1 - T_6} \tag{1}$$

For a closed cycle using an ideal gas as working fluid, recuperator effectiveness is expressed as

$$E = \frac{T_2 - T_3}{T_2 - T_5} = \frac{T_6 - T_5}{T_2 - T_5} \tag{2}$$

From equation (2), the following two relationships are obtained:

$$T_6 = T_2 - T_3 + T_5 \tag{2a}$$

and

$$T_3 = T_2 - E(T_2 - T_5) \tag{2b}$$

Substitution of equations (2a) and (2b) into equation (1) yields

$$\eta_{cy} = \frac{T_1 - T_2 + T_4 - T_5}{T_1 - ET_2 - (1 - E)T_5} \tag{3}$$

Division of both the numerator and the denominator by T_1 and expression of T_5/T_1 as $(T_5/T_4)(T_4/T_1)$ yields

$$\eta_{cy} = \frac{1 - \frac{T_2}{T_1} + \frac{T_4}{T_1}\left(1 - \frac{T_5}{T_4}\right)}{1 - E\frac{T_2}{T_1} - (1 - E)\frac{T_4}{T_1}\frac{T_5}{T_4}} \tag{4}$$

Equation (4) shows cycle efficiency to be a function of recuperator effectiveness and the following temperature ratios: turbine exit to inlet, compressor exit to inlet, and compressor inlet to turbine inlet.

Compressor temperature ratio (T_5/T_4) can be expressed as a function of turbine temperature ratio, turbine efficiency, compressor efficiency, and the pressure losses in the heat transfer components. The enthalpy change of the fluid in the compressor is

$$\Delta h_C = wc_p(T_5 - T_4) = wc_p \frac{(T_{5,id} - T_4)}{\eta_C} \tag{5}$$

Cancellation of wc_p and rearrangement of equation (5) yields

$$\frac{T_5}{T_4} = 1 + \frac{1}{\eta_C}\left(\frac{T_{5,id}}{T_4} - 1\right) \tag{5a}$$

The compressor pressure ratio can be expressed as

$$\frac{p_5}{p_4} = \left(\frac{p_2}{p_3}\frac{p_3}{p_4}\frac{p_5}{p_6}\frac{p_6}{p_1}\right)\frac{p_1}{p_2} = \left(\frac{1}{r_T/r_C}\right)\frac{p_1}{p_2} \tag{6}$$

As seen from equation (6), the factor r_T/r_C is equal to the ratio of turbine inlet to exit pressure divided by the ratio of compressor exit to inlet pressure and is also equal to the product of the ratios of exit to inlet pressure for all the heat transfer components. Consequently, r_T/r_C represents the fraction of compressor pressure ratio that can be recovered to do work in the turbine and is an indicator of the heat transfer component pressure drops. For the purpose of brevity, r_T/r_C will be subsequently referred to as the loss pressure ratio.

The isentropic state equation is

$$\frac{T_{5,id}}{T_4} = \left(\frac{p_5}{p_4}\right)^{(\gamma-1)/\gamma} \tag{7}$$

and substitution of equations (6) and (7) into equation (5a) yields

$$\frac{T_5}{T_4} = 1 + \frac{1}{\eta_C}\left[\left(\frac{1}{r_T/r_C}\right)^{(\gamma-1)/\gamma}\left(\frac{p_1}{p_2}\right)^{(\gamma-1)/\gamma} - 1\right] \tag{8}$$

The enthalpy change of the fluid in the turbine is

$$-\Delta h_T = wc_p(T_1 - T_2) = wc_p\eta_T(T_1 - T_{2,id}) \tag{9}$$

which upon simplification and rearrangement yields

$$\frac{T_{2,id}}{T_1} = 1 - \frac{1}{\eta_T}\left(1 - \frac{T_2}{T_1}\right) \tag{9a}$$

Substitution of equation (9a) into the isentropic state equation

$$\left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma} = \frac{T_{2,id}}{T_1} = 1 - \frac{1}{\eta_T}\left(1 - \frac{T_2}{T_1}\right) \tag{10}$$

and substitution of equation (10) into equation (8) yields

$$\frac{T_5}{T_4} = 1 + \frac{1}{\eta_C}\left[\frac{\left(\frac{1}{r_T/r_C}\right)^{(\gamma-1)/\gamma}}{1 - \frac{1}{\eta_T}\left(1 - \frac{T_2}{T_1}\right)} - 1\right] \tag{11}$$

Equation (11) is now substituted into equation (4) with a slight rearrangement to give the desired expression for cycle efficiency

$$\eta_{cy} = \frac{1 - \frac{T_2}{T_1} - \frac{T_4}{T_1}\left(\frac{1}{\eta_C}\right)\left[\frac{\left(\frac{r_T}{r_C}\right)^{(1-\gamma)/\gamma}}{1 - \frac{1}{\eta_T}\left(1 - \frac{T_2}{T_1}\right)} - 1\right]}{1 - E\frac{T_2}{T_1} - (1 - E)\frac{T_4}{T_1}\left\{1 + \frac{1}{\eta_C}\left[\frac{\left(\frac{r_T}{r_C}\right)^{(1-\gamma)/\gamma}}{1 - \frac{1}{\eta_T}\left(1 - \frac{T_2}{T_1}\right)} - 1\right]\right\}} \tag{12}$$

Fluid specific capacity rate, wc_p/P, which is another indicator of cycle performance, is required for the computation of specific radiator area and is herein derived. The required flow rate for the working fluid can be expressed as

$$w = \frac{\text{Net shaft power}}{\text{Net work per pound of fluid}}$$

which symbolically is

$$w = \frac{3415\ P}{c_p\left[(T_1 - T_2) - (T_5 - T_4)\right]} = \frac{3415\ P}{c_p T_1\left[1 - \frac{T_2}{T_1} - \frac{T_4}{T_1}\left(\frac{T_5}{T_4} - 1\right)\right]} \tag{13}$$

Substitution of equation (11) into equation (13) and rearrangement yields the desired expression for specific capacity rate.

$$\frac{wc_p}{P} = \frac{3415}{T_1\left\{1 - \frac{T_2}{T_1} - \frac{T_4}{T_1}\left(\frac{1}{\eta_C}\right)\left[\frac{\left(\frac{r_T}{r_C}\right)^{(1-\gamma)/\gamma}}{1 - \frac{1}{\eta_T}\left(1 - \frac{T_2}{T_1}\right)} - 1\right]\right\}} \tag{14}$$

The working fluid was assumed to be a monatomic gas with a specific heat ratio, γ, of 1.67

Radiator Area

The radiator is considered to be a tube or series of tubes either without fins or with fins of 100 percent effectiveness (no resistance to heat transfer). Consequently, all radiating area is prime area. The following assumptions are made for this computation:

(1) Sink temperature is constant for any given radiator. The sink temperature can be defined as the equilibrium temperature that a body in space will attain if there are no thermal influences other than the radiant heat

absorbed from and emitted to space. Sink temperature, consequently, depends on such controllable factors as the absorptivity-emissivity characteristics of the radiating surface and the orientation of the radiator with respect to the space radiant energy sources.

(2) The gas film heat transfer coefficient is constant throughout the radiator.

(3) The temperature drop through the tube wall is negligible.

(4) Heat conduction along the tube axis is neglected.

For an element of tube length, the heat transferred from the fluid to the tube wall must equal the heat radiated.

$$h_i(T - T_w)dA_i = \sigma\epsilon(T_w^4 - T_s^4)dA_R \tag{15}$$

A gas film heat transfer coefficient related to radiating area can be defined as

$$h_R = h_i \frac{dA_i}{dA_R} \tag{16}$$

Substituting equation (16) into equation (15) and solving for T yields

$$T = T_w + \frac{\sigma\epsilon}{h_R}(T_w^4 - T_s^4) \tag{17}$$

For the element of tube length under consideration, the decrease in fluid sensible heat must also equal the heat radiated.

$$-wc_p\ dT = \sigma\epsilon(T_w^4 - T_s^4)dA_R \tag{18}$$

Differentiation of equation (17) and substitution of the differentiated expression into equation (18) yields

$$-wc_p\left(\frac{4\sigma\epsilon}{h_R}T_w^3 + 1\right)dT_w = \sigma\epsilon(T_w^4 - T_s^4)dA_R \tag{19}$$

Rearrangement of equation (19) in order to separate the variables results in

$$dA_R = -wc_p \left[\frac{4T_w^3}{h_R(T_w^4 - T_s^4)} + \frac{1}{\sigma\epsilon(T_w^4 - T_s^4)} \right] dT_w \tag{20}$$

Dividing both sides of equation (20) by P and integrating between the limits of 0 to A_R and $T_{w,3}$ to $T_{w,4}$ yields

$$\frac{A_R}{P} = \frac{wc_p}{P} \left\{ \frac{1}{h_R} \ln \frac{T_{w,3}^4 - T_s^4}{T_{w,4}^4 - T_s^4} + \frac{1}{4\sigma\epsilon T_s^3} \left[\ln \frac{(T_{w,3} - T_s)(T_{w,4} + T_s)}{(T_{w,4} - T_s)(T_{w,3} + T_s)} - 2\left(\arctan \frac{T_{w,3}}{T_s} - \arctan \frac{T_{w,4}}{T_s} \right) \right] \right\} \tag{21}$$

where T_w is related to T by equation (17) and wc_p/P is obtained from equation (14).

RESULTS OF ANALYSIS

The developed equations showed that cycle efficiency and specific prime radiator area are functions of several system design factors and two independent temperature variables, turbine exit to inlet temperature ratio and compressor inlet to turbine inlet temperature ratio. Cycle efficiency, as seen from equation (12), depends on such design factors as turbine and compressor efficiencies, loss pressure ratio, and recuperator effectiveness. Specific prime radiator area, equation (21), also depends on the above mentioned design factors as well as the additional factors of turbine inlet temperature, sink temperature, radiating surface emissivity, and gas heat transfer coefficient.

The results of the analysis are discussed first with respect to the effects of the cycle temperature variables and design factors on cycle efficiency and then with respect to the effects of these same factors on specific radiator area. Except where otherwise indicated, the following set of design factors were used to compute the cycle efficiencies and prime radiator areas.

Turbine inlet temperature, $^{\circ}R$	2160
Sink temperature, $^{\circ}R$	400
Turbine efficiency	0.85
Compressor efficiency	0.80
Loss pressure ratio	0.90
Recuperator effectiveness	0.80
Radiator surface emissivity	0.86
Gas heat transfer coefficient, Btu/(hr)(sq ft rad. area)($^{\circ}R$)	5

Cycle Efficiency

As mentioned previously, cycle efficiency is a function of several system design factors and two temperature variables. For a given set of system design factors, cycle efficiency is shown in figure 2 plotted against compressor inlet to turbine inlet temperature ratio, T_4/T_1, for several values of turbine exit to inlet temperature ratio, T_2/T_1. The important things to note from figure 2 are the rapid decrease in cycle efficiency as T_4/T_1 increases, and the fact that at each value of T_4/T_1 there is one particular value of T_2/T_1 that maximizes cycle efficiency. As T_4/T_1 increases from 0.20 to 0.35, maximum attainable cycle efficiency decreases by more than a factor of two.

For a given set of cycle temperature variables, the change in cycle efficiency with turbomachinery efficiency, loss pressure ratio, and recuperator effectiveness are presented in figures 3(a), 3(b), and 3(c), respectively. It is seen from figures 3(a) and 3(b) that cycle efficiency rapidly deteriorates as the turbine and compressor efficiencies or the loss pressure ratio decrease. Reductions in the turbine and compressor efficiencies from 0.90 to 0.80 and 0.80 to 0.70 result in cycle efficiency decreasing by 30 and 65 percent, respectively, while similar reductions in loss pressure ratio cause cycle efficiency to decrease by 20 and 30 percent, respectively. The need for high turbomachinery efficiency and low pressure drops in a Brayton cycle power system is clearly evident. Figure 3(c) shows that cycle efficiency can be significantly increased by increasing recuperator effectiveness and, for the typical case presented, an effectiveness of 0.86 results in a cycle efficiency double that obtainable without a recuperator (E = 0). The increase in cycle efficiency with increasing effectiveness occurs because as more heat is supplied to the gas in the recuperator, less heat must be supplied by the heat source. Recuperator weight, of course, is increasing with effectiveness and approaches infinity as effectiveness approaches one. The choice of a design effectiveness depends to a great extent on total system weight and more will be said about this in the radiator area discussion.

Radiator Area

Radiator area, too, is a function of several system design factors and two temperature variables. As mentioned previously, the computed radiator areas are prime areas which can be used to (1) serve as a guide for the selection of a desirable set of cycle temperature variables, and (2) show how the design factors affect both radiator area and the choice of temperature variables.

Radiator area is plotted against T_4/T_1 in figure 4 for several values of T_2/T_1. Examination of figure 4 shows that: (1) for each value of T_2/T_1 there results a curve which has a minimum radiator area at some value of T_4/T_1, (2) for each value of T_4/T_1 there is one value of T_2/T_1 that yields a minimum radiator area, and (3) an envelope curve (the dashed curve in fig. 4) drawn around the family of curves also shows a minimum radiator area. The shape of these curves is readily explainable. There are two opposing factors which affect radiator area. Radiative heat flux is proportional to the fourth power of temperature; consequently, an increase in radiator temperature level will act to reduce radiator area. However, as seen from figure 2, cycle efficiency decreases with an increase in radiator temperature level; consequently, there is an increase in radiator heat load which will act to increase radiator area. It is the interaction of these two effects which cause the minimum observed in figure 4. As seen from figure 4, the choice of cycle temperature variables must be restricted to those combinations yielding radiator areas in the vicinity of the minimum if the inherently large radiator is not to become even larger.

The discussion to follow will show how the design factors affect both radiator area and the choice of temperature variables. Envelope curves, similar to that shown in figure 4, will be used to represent radiator area. The value of T_2/T_1 which minimizes radiator area for any given T_4/T_1 will be subsequently called $(T_2/T_1)_{opt}$, and the $(T_2/T_1)_{opt}$ loci will be shown on the envelope curves.

The design factors which influence radiator area are turbine inlet temperature, sink temperature, turbomachinery efficiency, loss pressure ratio, recuperator effectiveness, gas heat transfer coefficient, and surface emissivity. The effect of turbine inlet temperature on radiator area is shown in figure 5, where prime radiator area is plotted against T_4/T_1 for turbine inlet temperatures of 1710°, 2160°, and 2500° R. Increasing turbine inlet temperature from 1710° to 2500° R reduces radiator area by a factor of nearly four. Such an area reduction is very significant since, as can be seen from figure 5, hundreds or thousands of square feet of radiator area, depending on turbine inlet temperature and system power level, will be required for Brayton cycle power systems. The reduction in radiator area with increasing turbine inlet temperature is due to the increase in radiative heat flux which is proportional to the fourth power of radiator temperature. As turbine inlet temperature increases, there is a small decrease in the optimum values of T_4/T_1 and T_2/T_1. In addition, the choice of T_4/T_1 becomes less restricted as the temperature level increases.

The effect of sink temperature on radiator area is shown in figure 6, where prime radiator area is plotted against T_4/T_1 for sink temperatures of 0°, 400°, and 600° R. Radiator area increases with increasing sink temperature due to a reduction in the net radiative heat flux; the increase in area becomes quite significant as sink temperature begins to approach compressor inlet temperature. For the case shown in figure 6, an increase in sink temperature from 0° to 400° R results in a 13 percent increase in radiator area while further increase from 400° to 600° R results in a 30 percent increase in radiator area. The optimum values of T_4/T_1 and T_2/T_1 increase slightly and their proper choice becomes more critical as sink temperature increases.

The combined effects of turbine inlet temperature and sink temperature on radiator area are shown in figure 7, where minimum prime radiator area is plotted against turbine inlet temperature for several values of sink temperature. The radiator areas presented in this figure are the minimum obtainable (i.e., those areas corresponding to the optimum values of T_2/T_1 and T_4/T_1) for each combination of turbine inlet and sink temperatures. As was seen from the two previous figures, radiator area decreases with increasing turbine inlet temperature and decreasing sink temperature. The effect of sink temperature on radiator area is seen to become more significant as turbine inlet temperature decreases. Operation at higher turbine inlet temperatures, aside from reducing radiator area, has the advantage of lessening operating fluctuations due to a change in sink temperature with position in space.

The effect of turbomachinery efficiency on radiator area is shown in figure 8, where prime radiator area is plotted against T_4/T_1 for turbine and compressor efficiencies of 0.70, 0.80, and 0.90. A reduction in turbomachinery efficiency results in a very significant increase in radiator area. As seen from figure 8, a reduction in turbine and compressor efficiencies from 0.90 to 0.80 results in a twofold increase in radiator area while a further reduction from 0.80 to 0.70 causes an additional threefold increase in radiator area. This increase in radiator area is due primarily to the decrease in cycle efficiency shown in figure 3(a). The optimum value of T_4/T_1, as seen from figure 8, decreases with decreasing turbomachinery efficiency in order to offset the rapid deterioration of cycle efficiency;

consequently, radiator temperature has decreased, and the combined effects of lower radiator temperature and cycle efficiency cause the large increase in radiator area. The optimum value of T_2/T_1 is seen to increase with a reduction in turbomachinery efficiency.

The effect of loss pressure ratio on radiator area is shown in figure 9, where prime radiator area is plotted against T_4/T_1 for loss pressure ratios of 0.70, 0.80, and 0.90. A reduction in loss pressure ratio results in a large increase in radiator area. As seen from figure 9, a reduction in loss pressure ratio from 0.90 to 0.80 results in a 50 percent increase in radiator area while a further reduction from 0.80 to 0.70 causes an additional 55 percent increase in radiator area. The increase in radiator area and decrease in optimum T_4/T_1 with decreasing loss pressure ratio occur for the same reasons as explained above for decreasing turbomachinery efficiency. The optimum value of T_2/T_1 decreases with a reduction in loss pressure ratio.

The effect of recuperator effectiveness on radiator area is shown in figure 10, where prime radiator area is plotted against T_4/T_1 for recuperator effectivenesses of 0, 0.5, and 1. Radiator area decreases with increasing effectiveness, and the area required for $E = 1$ is about 70 percent of that required for $E = 0$. The reduction in radiator area with increasing effectiveness is due to the increase in cycle efficiency, as shown in figure 3(c). Although cycle efficiency is more than doubled as effectiveness increases from 0 to 1, the decrease in radiator area is not proportionately as great because the area reduction occurs at the high temperature (most efficient) end of the radiator. Due to the inherently large size of the radiator, even a 20 or 25 percent reduction in radiator area, as can be achieved with recuperator effectivenesses of 0.8 to 0.9, can result in enough of a savings to offset the additional weight and pressure drop attributable to a recuperator. The optimum value of T_4/T_1 increases only slightly with increasing effectiveness; the optimum value of T_2/T_1, however, increases significantly as effectiveness increases. This increase in T_2/T_1 is advantageous since it results in a reduction in the pressure ratio requirement for the turbomachinery.

The effect of gas heat transfer coefficient on radiator area is shown in figure 11, where prime radiator area is plotted against T_4/T_1 for gas heat transfer coefficients of 5, 20, and infinity. The heat transfer coefficient presented in this figure is the coefficient relative to radiating area which, according to equation (16), is equal to the coefficient relative to internal heat transfer area times the ratio of internal heat transfer area to radiating area. For meteoroid protected tube and fin radiators, this area ratio can be as low as 0.10 to 0.25 and, consequently, cause the heat transfer coefficient relative to radiating area to be quite low. Radiator area increases with decreasing heat transfer coefficient; however, as seen from figure 11, the effect of heat transfer coefficient on radiator area is quite small for coefficients in excess of about 20, but becomes significant as the heat transfer coefficient decreases below 20. A reduction in heat transfer coefficient from 20 to 5 results in a 40 percent increase in radiator area. For those cases where the heat transfer coefficient is quite low, the use of internal fins, which greatly increase the ratio of internal to radiating area, can be beneficial. The optimum values of T_4/T_1 and T_2/T_1 increase slightly with increasing heat transfer coefficient.

The effect of radiator surface emissivity on radiator area is shown in figure 12, where prime area is plotted against T_4/T_1 for emissivities of 0.6, 0.8, and 1.0. Radiator area increases with decreasing emissivity since the radiative heat flux is directly proportional to emissivity. A reduction in emissivity from 1.0 to 0.8 results in an 11 percent increase in radiator area while a further reduction from 0.80 to 0.60 causes an additional 23 percent increase in area. The optimum values of T_4/T_1 and T_2/T_1 appear to be nearly independent of emissivity.

It is interesting to note that the optimum values of T_2/T_1 and T_4/T_1, even over the wide range of design factors investigated, were generally in the range of 0.70 to 0.80 and 0.25 to 0.35, respectively.

SUMMARY OF RESULTS

This analysis was conducted in order to obtain an understanding of the thermodynamic characteristics of Brayton cycles for space application. The characteristics of interest are system performance, as denoted by cycle efficiency, and a desirable set of cycle temperatures. Since the radiator is the largest component and a major weight contributor to the system, minimum prime radiator area was selected as the criterion for cycle temperature selection.

Cycle efficiency and prime radiator area are functions of several system design factors and two independent temperature variables. Cycle efficiency depends on such design factors as turbine and compressor efficiencies, loss pressure ratio, and recuperator effectiveness. Radiator area also depends on the above mentioned design factors as well as the additional factors of turbine inlet temperature, sink temperature, radiating surface emissivity, and gas heat transfer coefficient. The two independent temperature variables were turbine exit to inlet temperature ratio and compressor inlet to turbine inlet temperature ratio.

At each ratio of compressor inlet to turbine inlet temperature there is one particular value of turbine exit to inlet temperature ratio that maximizes cycle efficiency and an increase in compressor inlet to turbine inlet temperature ratio results in a decrease in this maximum cycle efficiency. Decreases in turbine and compressor efficiencies and loss pressure ratio result in a rapid deterioration of cycle efficiency. The use of a recuperator offers a potential twofold increase in cycle efficiency.

For any given set of design factors, a study of radiator area reveals that: (1) for each value of the ratio of turbine exit to inlet temperature there is one value of compressor inlet to turbine inlet temperature ratio that yields a minimum radiator area; (2) for each value of the ratio of compressor inlet to turbine inlet temperature there is one value of turbine exit to inlet temperature ratio that yields a minimum radiator area; and (3) there is one combination of the two temperature variables that yields a minimum radiator area with respect to both variables. Required radiator area can be reduced by increasing turbine inlet temperature, turbomachinery efficiency, loss pressure ratio, recuperator effectiveness, gas heat transfer coefficient, and surface emissivity and decreasing sink temperature.

The optimum values of the cycle temperature variables depend upon the particular values of the design factors. For design factors in the range of those usually encountered in a system design, the optimum values of turbine exit to inlet temperature ratio and compressor inlet to turbine inlet temperature ratio are generally in the range of 0.70 to 0.80 and 0.25 to 0.35, respectively.

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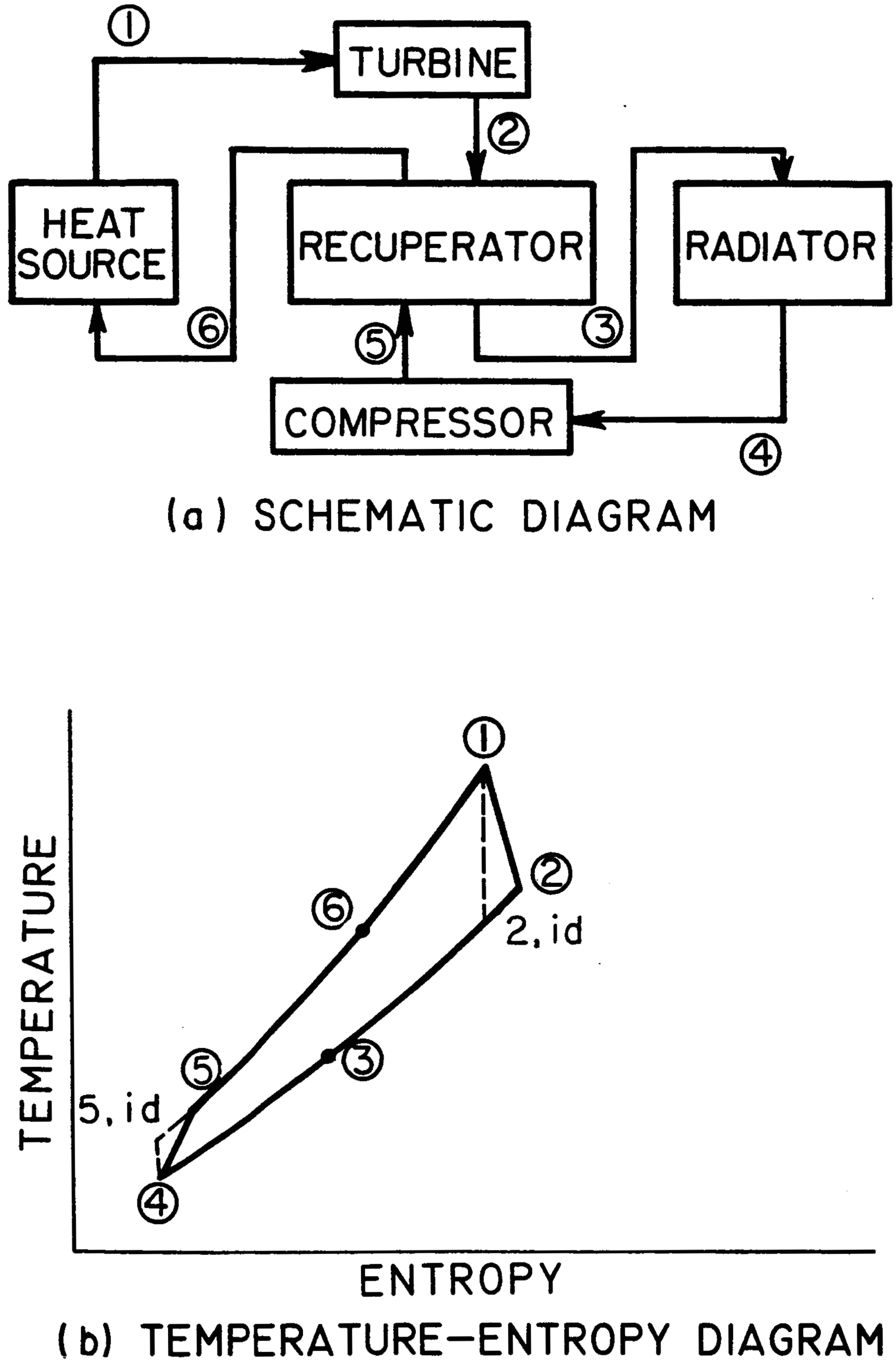

Figure 1. - Brayton power cycle.

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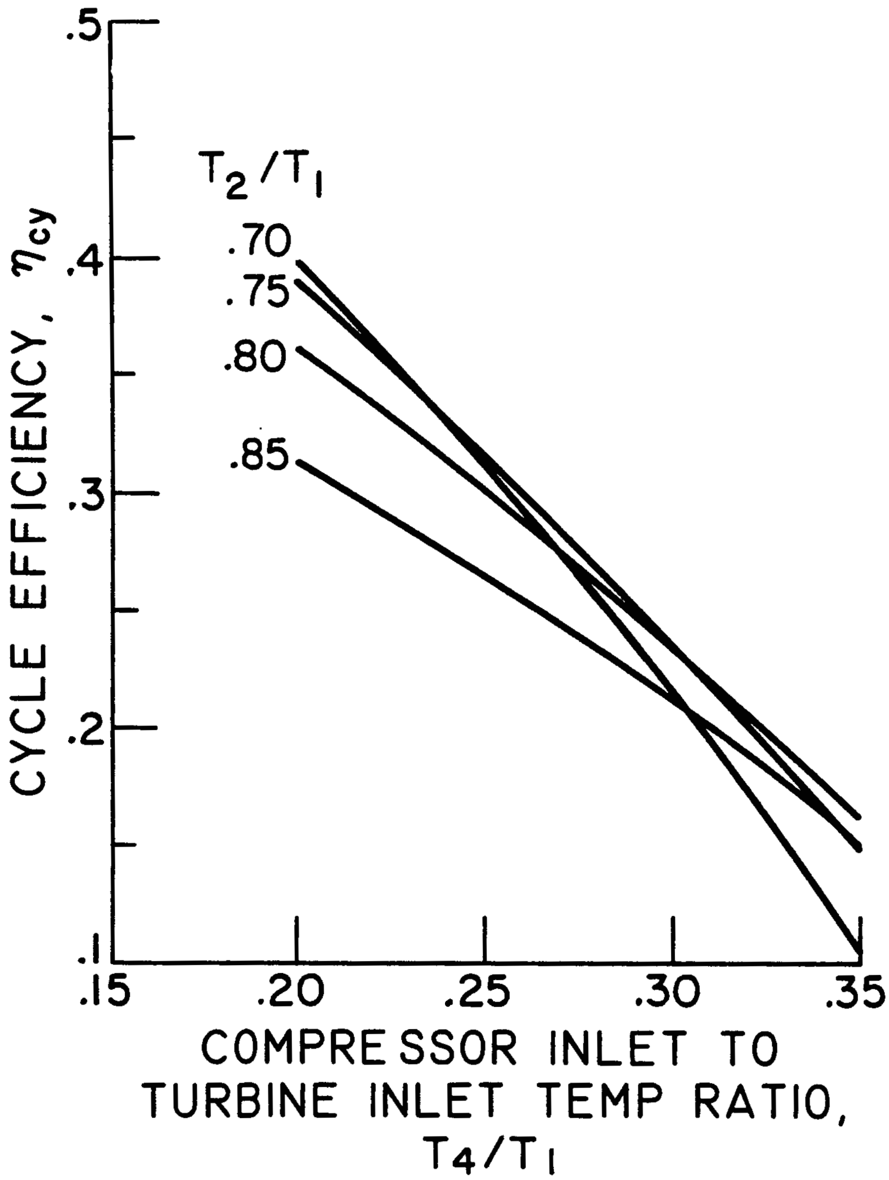

Figure 2. - Effect of cycle temperature variables on cycle efficiency.

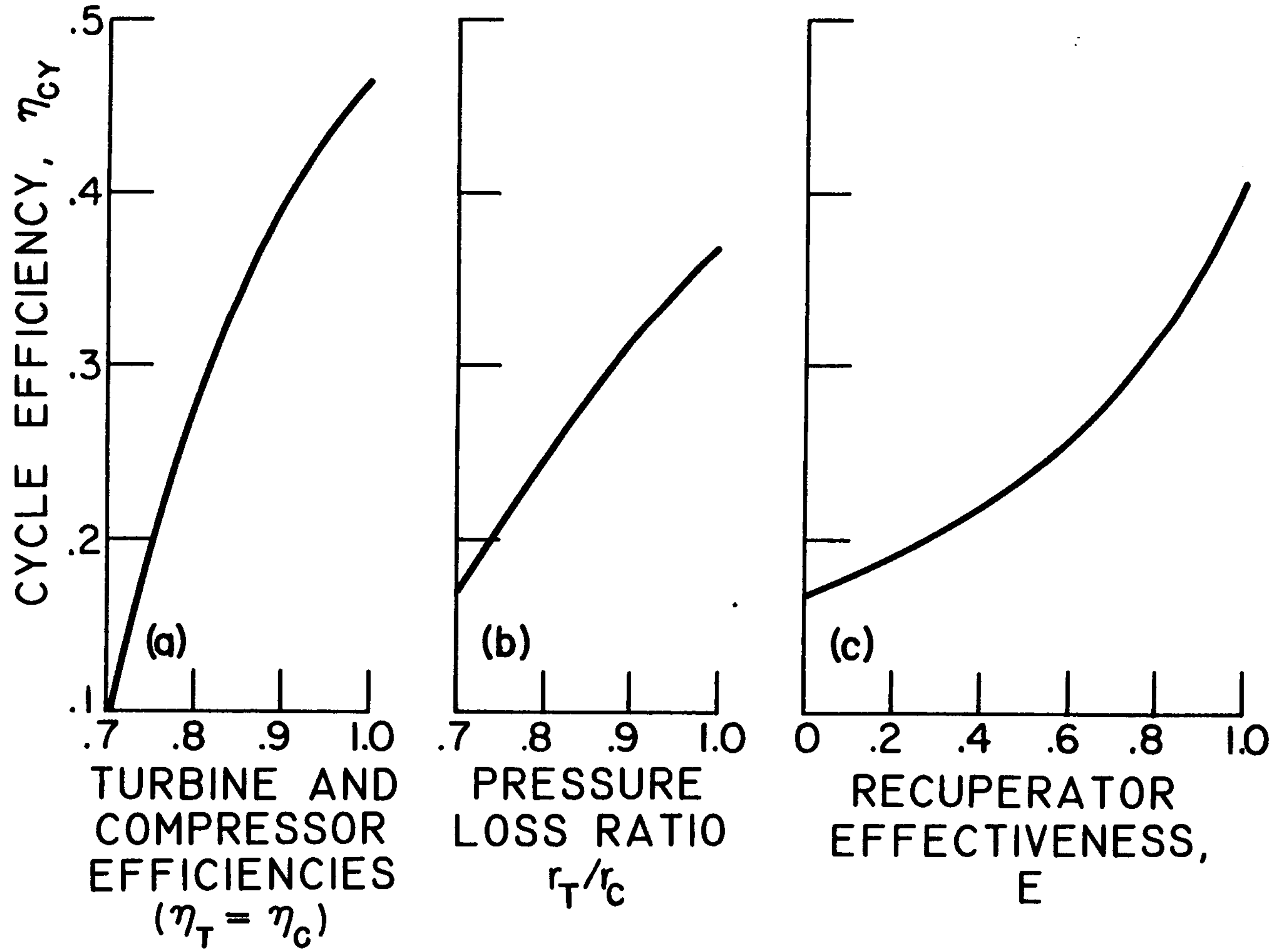

Figure 3. - Effect of system design factors on cycle efficiency. ($T_2/T_1 = 0.75$, $T_4/T_1 = 0.25$)

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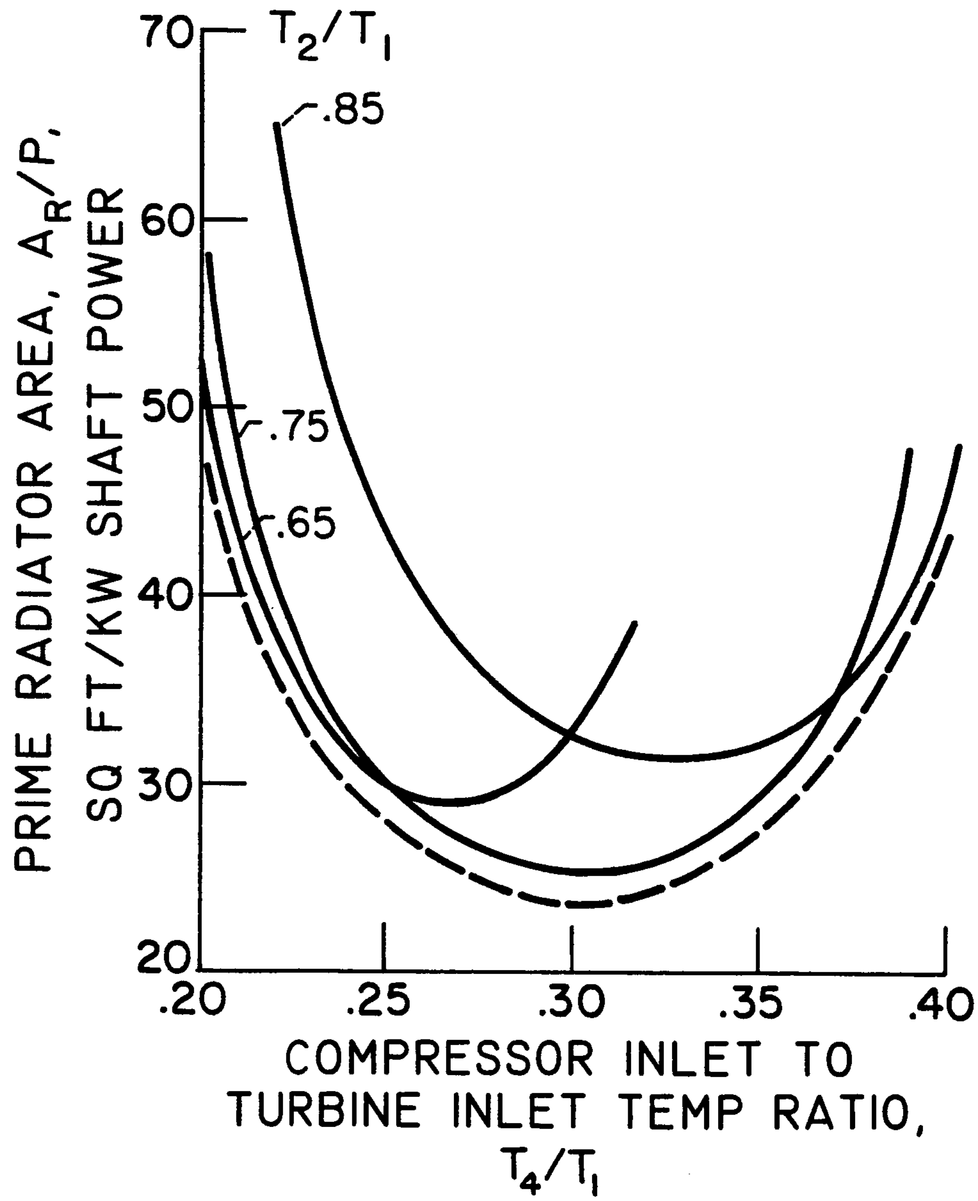

Figure 4. - Effect of cycle temperature variables on prime radiator area.

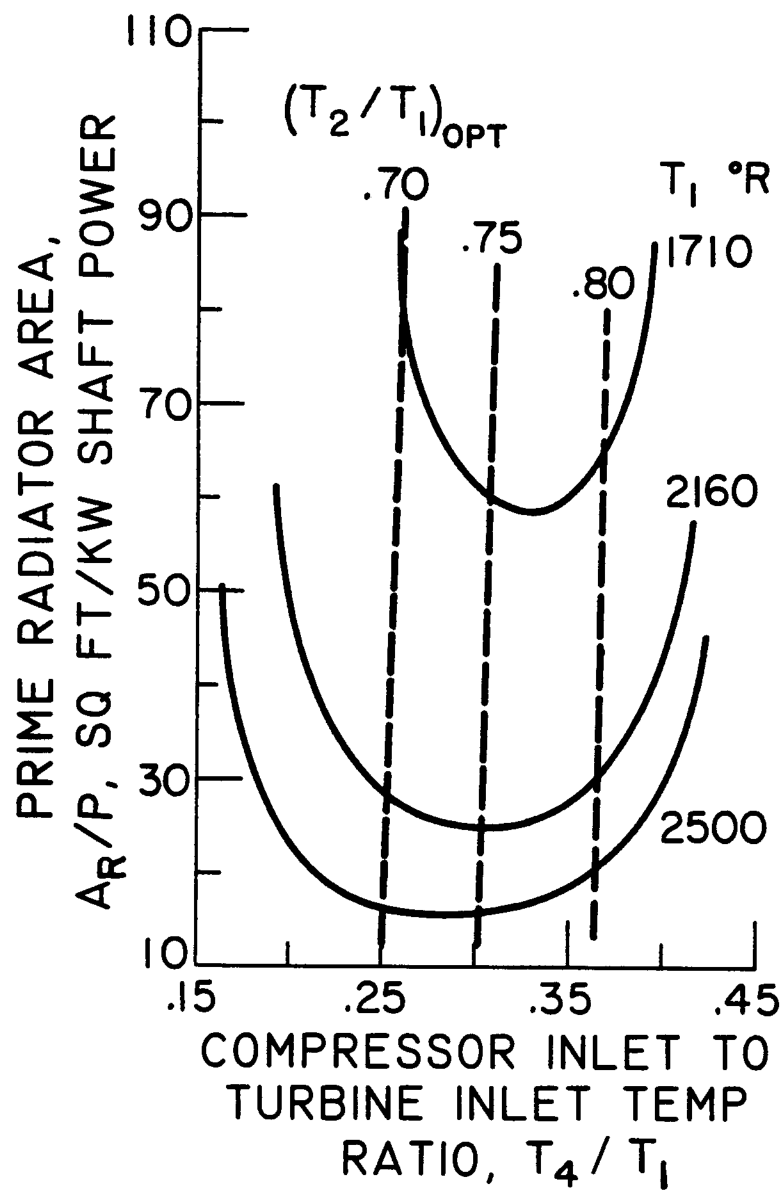

Figure 5. - Effect of turbine inlet temperature on prime radiator area.

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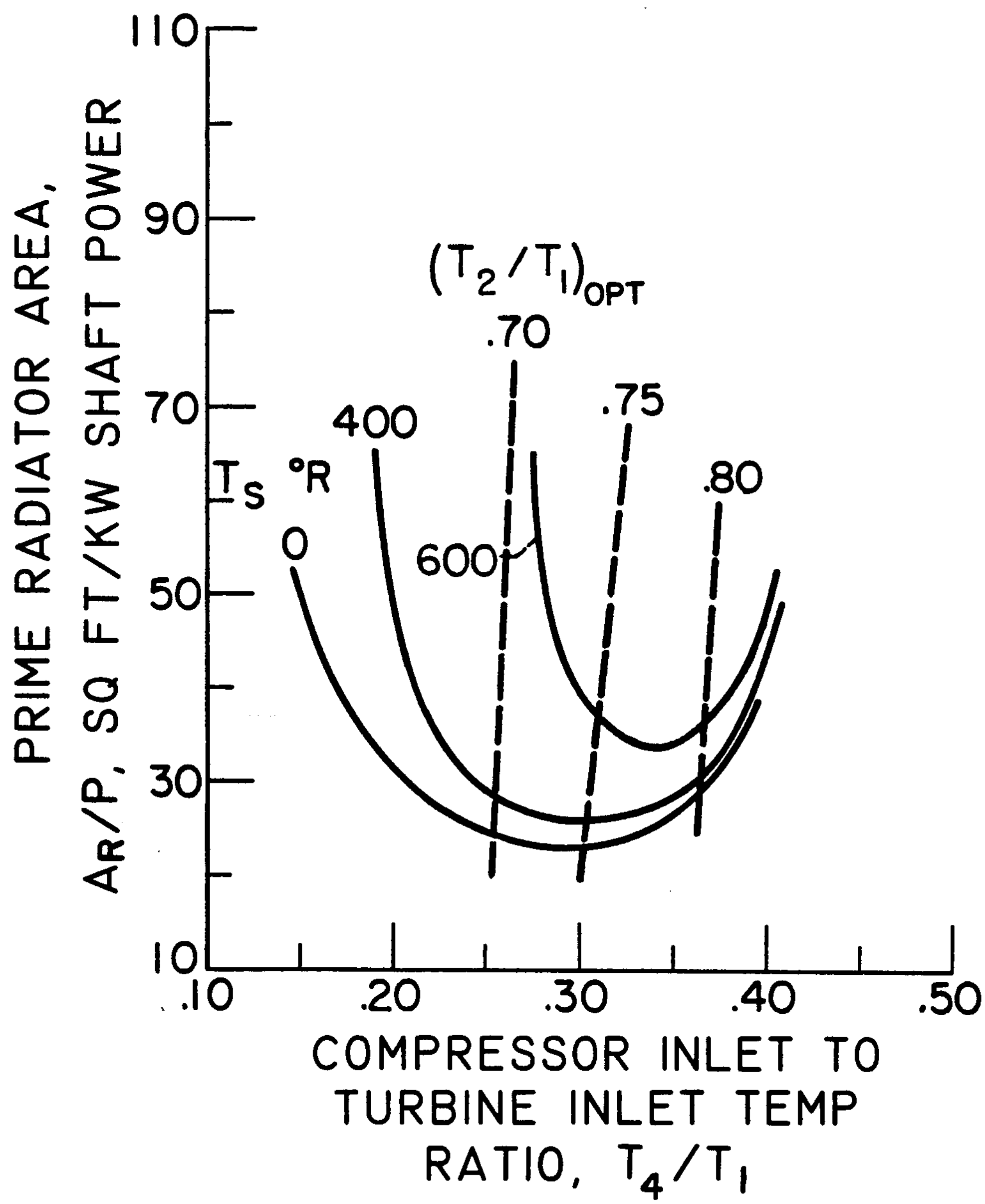

Figure 6. - Effect of sink temperature on prime radiator area.

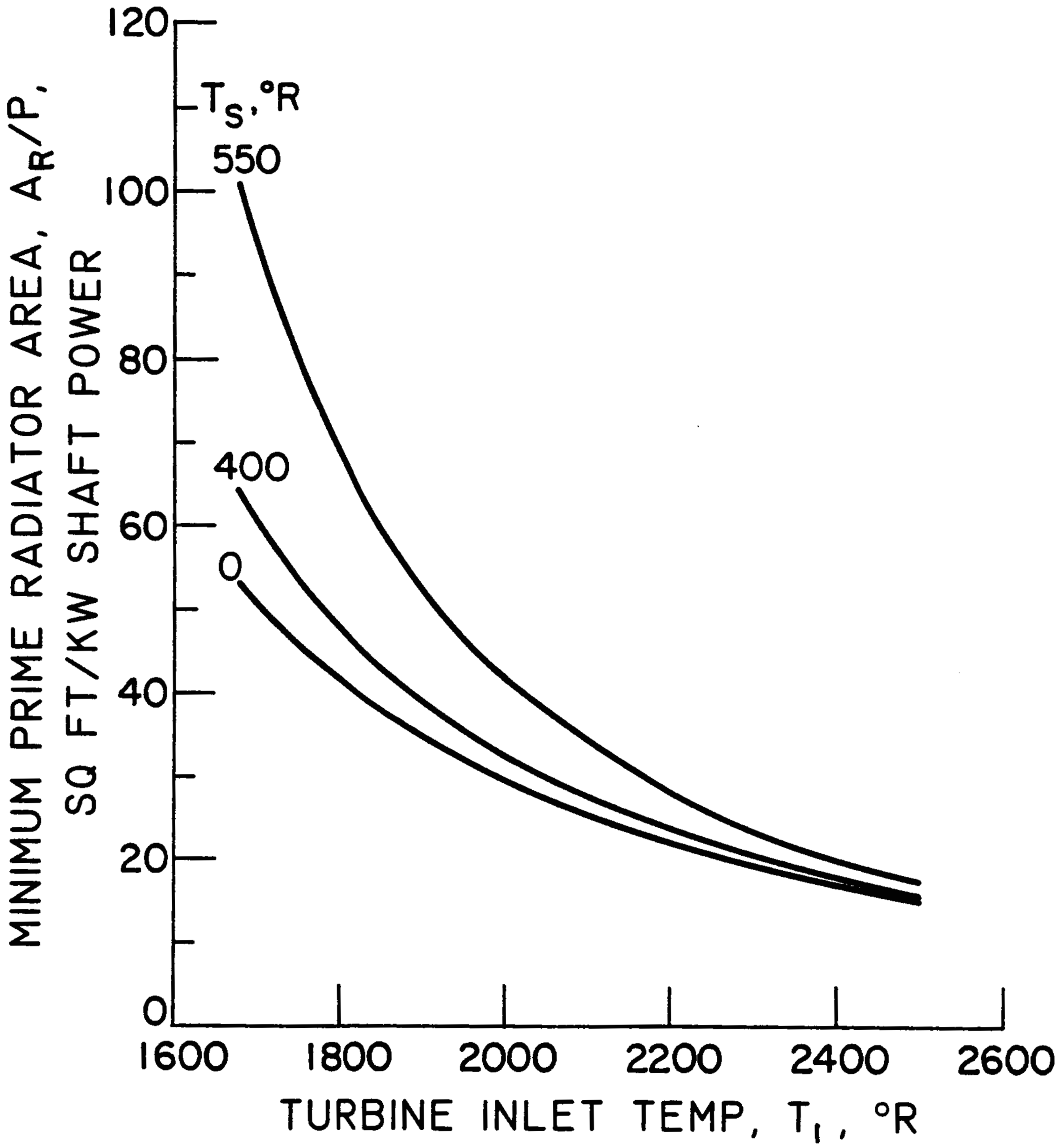

Figure 7. - Effect of turbine inlet and sink temperatures on prime radiator area.

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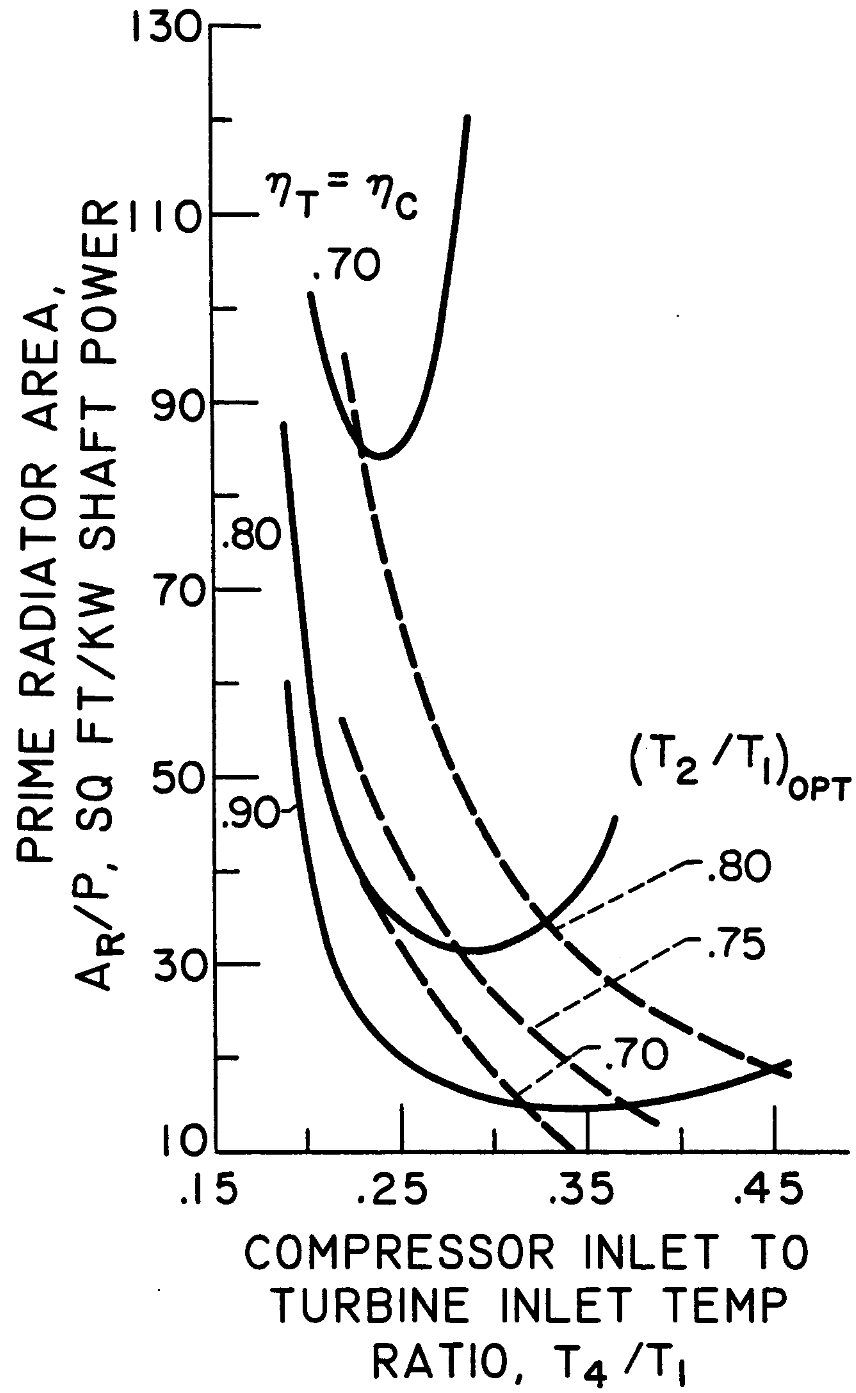

Figure 8. - Effect of turbomachinery efficiency on prime radiator area.

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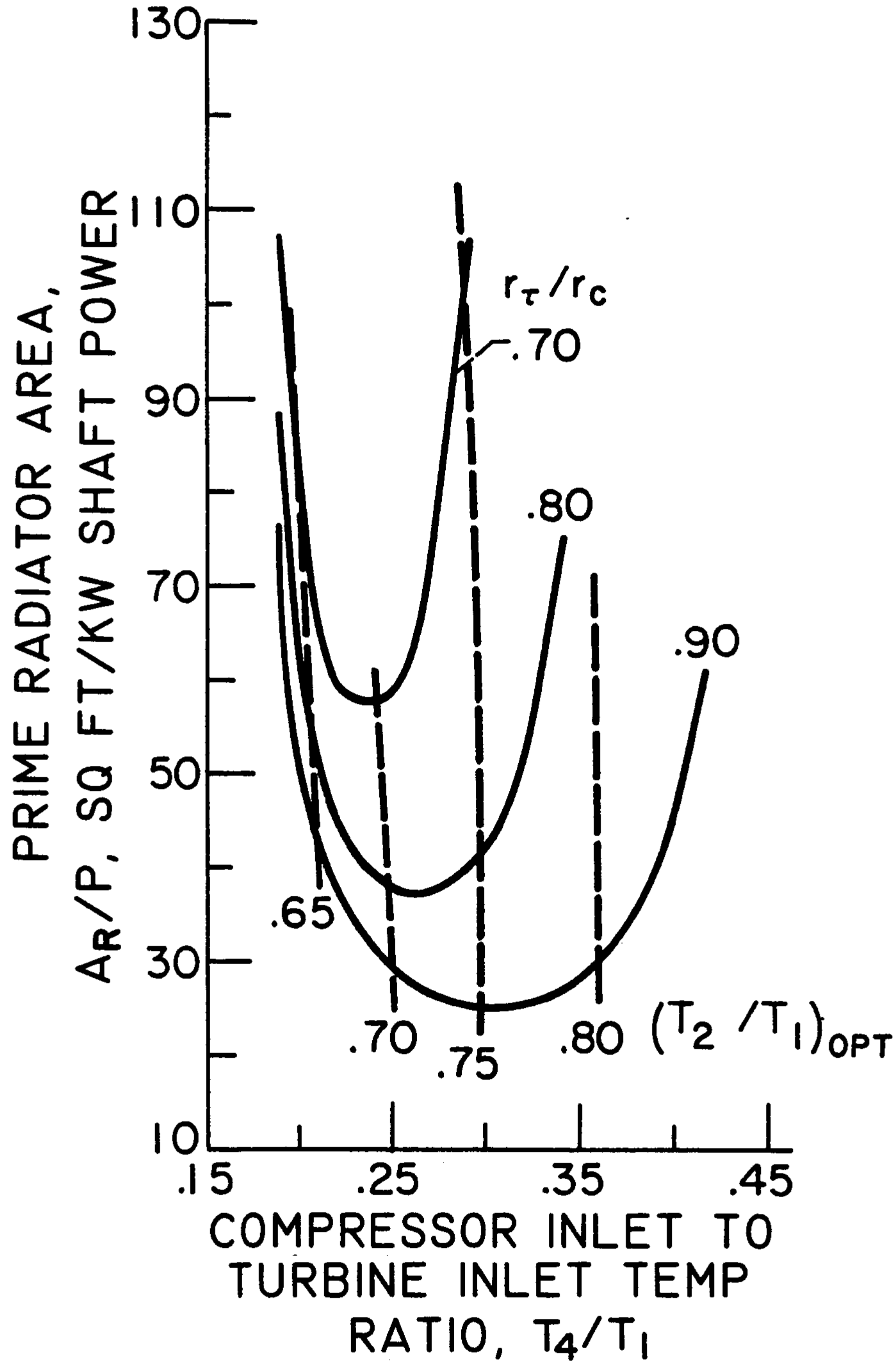

Figure 9. - Effect of pressure loss ratio on prime radiator area.

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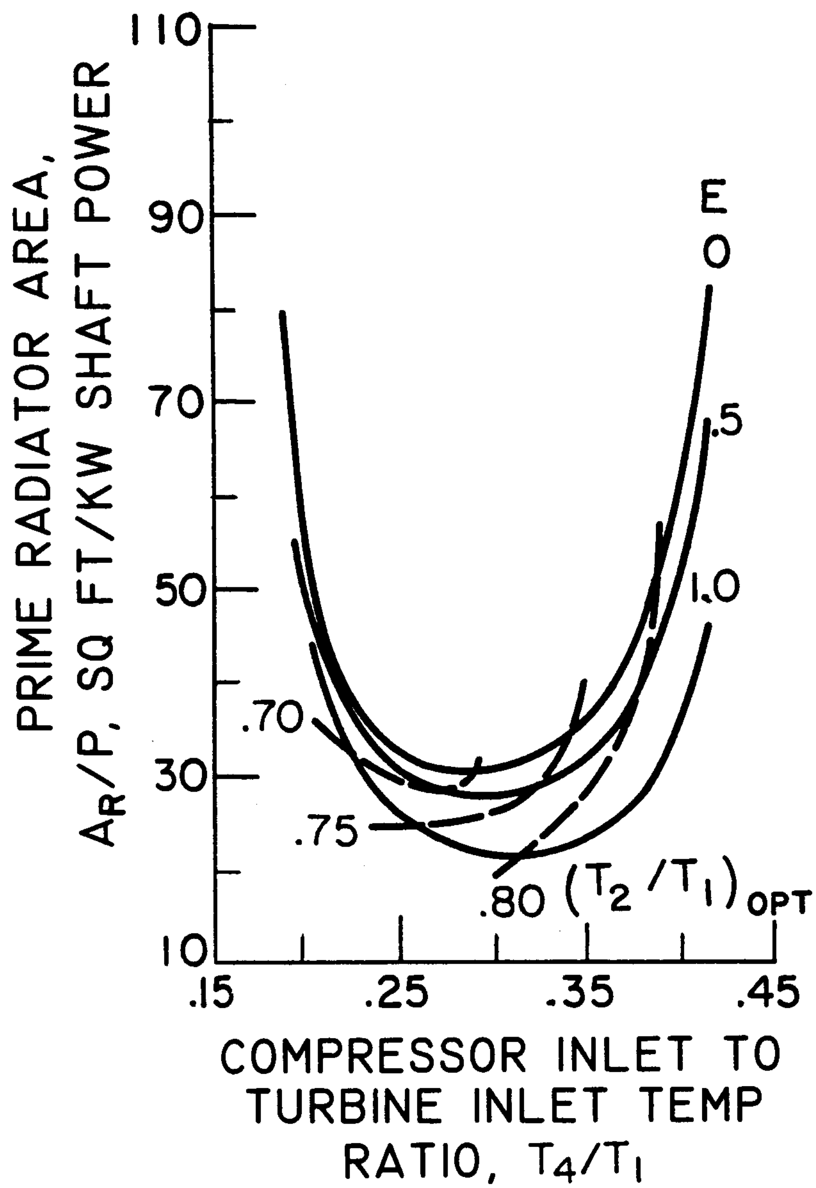

Figure 10. - Effect of recuperator effectiveness on prime radiator area.

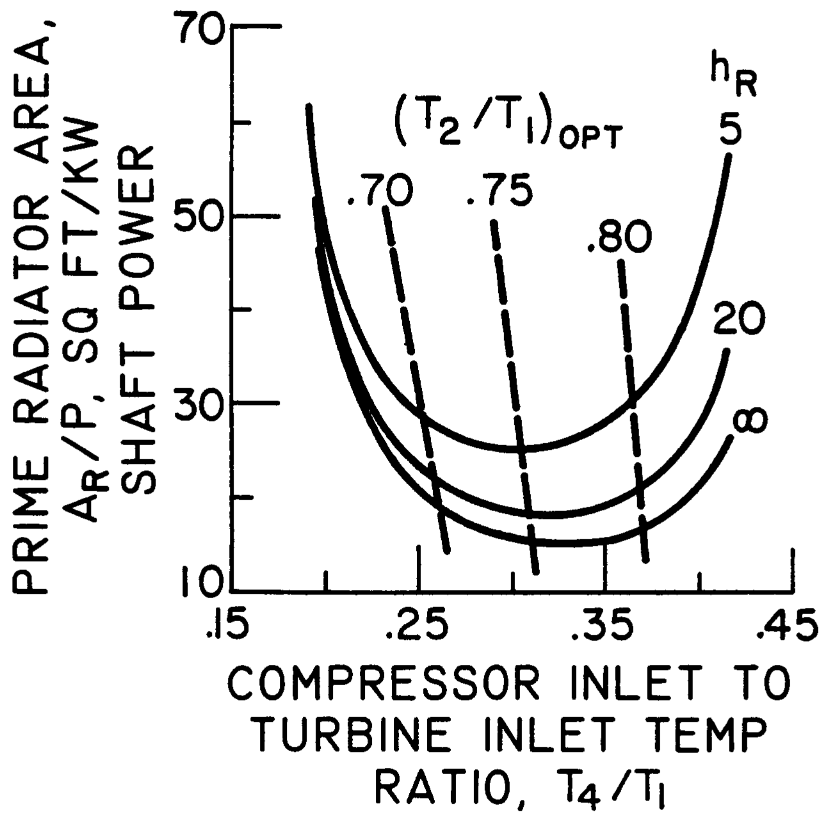

Figure 11. - Effect of gas heat transfer coefficient on prime radiator area.

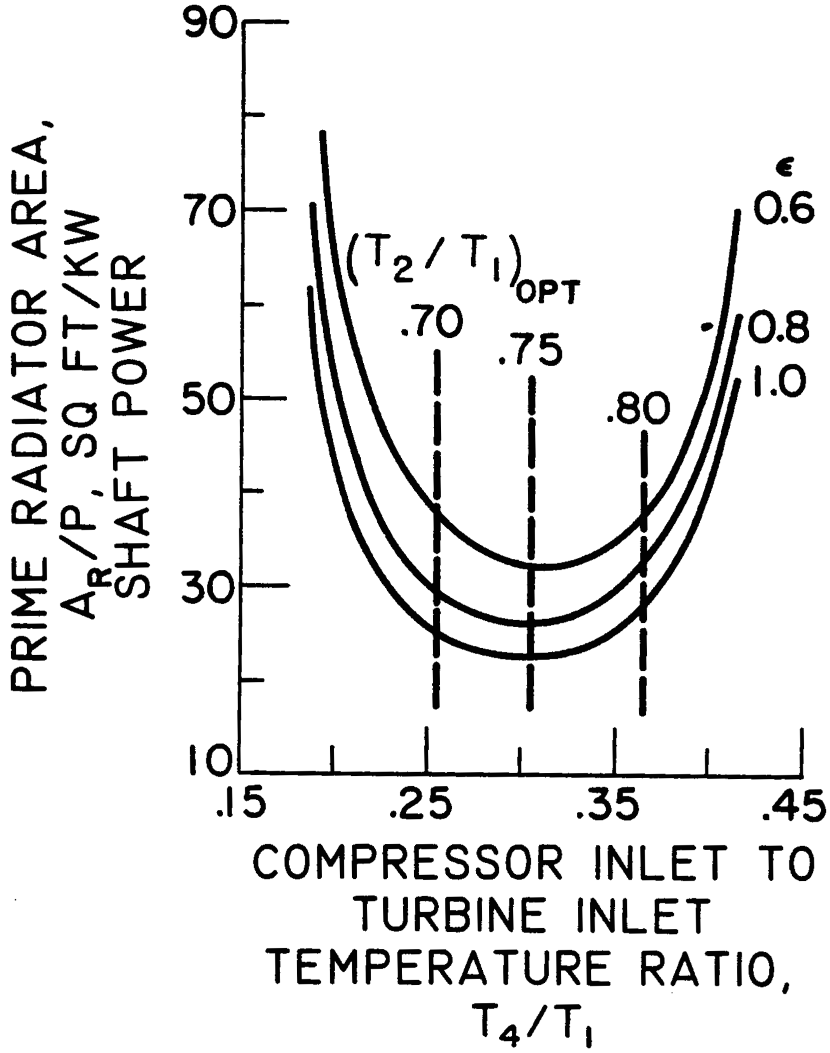

Figure 12. - Effect of radiator surface emissivity on prime radiator area.